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ADAPTING HERMETICALLY SEALED COMPRESSOR TECHNOLOGY TO DEAL WITH SOUR AND CORROSIVE GASES

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ABSTRACT

The Oil & Gas industry faces increasing challenges to meet growing energy demands with the most sustainable exploration and production techniques. Compressor manufacturers have responded to these challenges with the development of hermetically sealed compressors, enabling gas compression with minimized environmental footprint. The main feature in all of these designs is integration of driver and driven unit into a single casing. As a result, there is no shaft protruding out of the casing, hence there is no need for shaft sealing elements like dry gas seals that have traditionally suffered poorest reliability. There is one major disadvantage of this design concept: electrical components from electrical motor and active magnetic bearings are introduced in a process gas environment. This may not be so detrimental on compressor applications handling clean, export-quality gas allowing hermetically sealed technology with the electrical insulation system directly exposed to the process gas used for cooling purposes. However, for non-clean, toxic or corrosive applications the electrical components require either special treatment or, alternatively, need to be separated from the process gas. Authors' company has introduced the STC-ECO following the latter principle, a field-proven hermetically sealed, canned, compressor specifically designed to meet the requirements of the most demanding upstream applications. The applications have been defined and described in four distinct material categories.

The primary design concept of the Electrical, Canned and Oil-free (ECO) unit is its 'canned' technology. The electric motor, Active Magnetic Bearings and instrumentation are separated from the process gas by cylindrical parts integrated in stationary housing parts, such that no electrical component is exposed to process gas. A first prototype of a canned hermetically sealed compressor has operated over 50.000 hours since installation in 2006 on a natural gas asset in the Netherlands. For more severe sour and corrosive applications, special precautions were required on the compressor package design, among which a patented hybrid rotor design and hybrid thrust disk design. The hybrid technology contains separate sections of different materials welded together to form one multi-functional solid component. Furthermore, the motor stator 'can', separating the motor stator windings from the cooling process gas, was subjected to a series of qualification tests at research institutes in the Netherlands to qualify the compatibility to H_2S and corrosive conditions.

For an upstream gas compression project in the Kingdom of Saudi Arabia, such a 5.8 MW unit capable of handling sour gas was designed, built and ASME PTC-10 Type-1 tested at the Author's company facility in Hengelo, The Netherlands. Upon completion of an extensive test program including emergency landing test on the auxiliary bearing system, the unit was shipped and commissioned at site in the Middle East and started operation in June 2015.

The paper covers the design concept of the 'canned' hermetically sealed technology, highlights the design aspects making it suitable for sour and corrosive applications and describes the unit built and tested for the upstream gas compression project of Saudi Aramco.

INTRODUCTION

The upstream Oil-Gas separation and compression Project is located in a desert remote area at an upstream production unit in the Eastern Province of the Kingdom of Saudi Arabia. The project was officially launched as operational in July 2015. The design of the project includes a comprehensive monitoring and surveillance plan.

The main challenge to the Project team was to provide a modular, minimum-utility, easily relocatable compression solution that can handle corrosive gases with a very wide variation in gas Molecular Weight, flow and pressure. Safety was also a prime objective of the installation facilities. It was obvious that the hermetically sealed design fulfills most of the project requirements but needs to be advanced and extended to handle sour gases. The outcome was the world's 1st modular seal-less compressor designed for wet corrosive gas applications. By eliminating dry gas seals and lubricated bearings, along with their supporting auxiliary systems, the operational safety and reliability of the compressor are boosted due to elimination of a major source of leakage and unscheduled shutdowns. Moreover, flaring and consumption of utilities (other than electric power) are nearly eliminated. Benefits also include around 80% reduction in instrumentation on this specific installation and more than 50% reduction in installed space compared to a traditional compressor design. The inherent modular design of the integrated motor-compressor also allows easy installation, commissioning and relocation whenever necessary.

DESIGN CHALLENGES

The process gas is saturated hydrocarbon gas separated from crude oil. Its composition varies significantly over the plant's production life, and expected to contain considerable amounts of wet CO_2 in combination with H_2S and water vapor. This means that materials used for the motor-compressor have to withstand corrosive gas. The hermetically sealed compressor contains a single shaft for both the motor and compressor section, hence the rotor and thrust disk need to be manufactured from corrosion-resistant materials. However, a major draw-back of these materials is their low iron-content, hence they are not suitable for an efficient electric motor rotor and magnetic bearing thrust disk. In addition, the electrical insulation systems of motor stator windings and active magnetic bearing stator windings have to withstand the corrosive attack whilst allowing cooling media to extract heat generated because of electrical losses.

Therefore, the major design challenges for upstream gas compression applications are condensed to the following questions:

- How to solve the two contradictory functions of an hermetically sealed rotor and thrust disk: sour and corrosion resistant & good magnetic properties?
- How to solve the two contradictory functions of the separation 'can' of the motor stator: sour and corrosion resistant & non-conductive?

DEVELOPMENT APPROACH

In order to distinguish clearly defined areas of application and their associated material selection, a category definition was set focusing on application temperature, H_2S content, CO_2 partial pressure and the amount of Chlorides and gas humidity, see Table 1. Upstream gas process parameters clearly mandate material selections for Category 3 'Sour and Corrosive'. This category is more complicated as opposed to Categories 1 and 2 and dictates the use of advanced designs for the shaft and thrust disk.

Table 1: Material Categories for Hermetically sealed applications

	Category 1	**Category 2**	**Category 3**	**Category 4**
Designation	*Regular*	*Sour*	*Sour & Corrosive*	*Corrosive & High Cl^-*
Min Temp	-20 ºC	-20 ºC	-20 ºC	-20 ºC
Max Temp	150 (180) ºC	150 (180) ºC	150 (180) ºC	150 (180) ºC
H_2S Partial Pressure	< 0.0034 bar	0.0034 – 1 bar	0.0034 – 1 bar	1 – 10 bar
CO_2 Partial Pressure	< 1 bar (any, if gas is dry)	< 1 bar	1- 25 bar	Any
Chloride ions	<= 50 ppm	<=50 ppm	<=50 ppm	<=30,000 ppm
Humidity	Up to 100% Wet and Dry	Up to 100% Wet and Dry	Up to 100% Wet and Dry	Up to 100% Wet and Dry

The electrical insulation properties of the gas depend on the composition and pressure. It is well known that water vapor has harmful effect on the insulation properties. Gas molecules which will dissociate easily forming ions are also harmful to electrical properties. It is therefore not advisable to allow process gas to enter the stator windings. Further, the corrosion resistance of copper is a function of fluid velocity. Corrosive attack is accelerated by dissolved oxygen, carbon dioxide and or ammonia. Copper and copper based alloys are resistant to neutral and slightly alkaline solutions, dry gases, natural gas, and most other hydrocarbons. They are attacked by hydrogen sulfide and other sulfur compounds, most acids and strong alkalis, ref (4), (5). The process gas in the upstream application contains H_2S which will react with copper bars of the rotor as well as copper windings in the stator. The stator copper windings are insulated using insulating material. However it is not impermeable to gas and susceptible to high levels of corrosion with H_2S. Effect of H_2S and CO_2 on high voltage insulation materials was extensively studied by Sihvo, ref (5). His findings were that PET film insulation cannot withstand raw natural gas. Insulations based on epoxy resins reacted with water vapor over a period of time leading to degradation. H_2S also contributed to this degradation. All these factors contribute to the primary concept to avoid electrical windings being exposed to process gas for upstream applications. This also meant that the stator windings could not be cooled by circulation of process gas. And as there should be no need for external clean gas supply, the stator windings are cooled with a liquid in a closed loop circuit.

Separating the electrical motor stator windings from the process gas is achieved by a specially designed thin-walled 'can' in the air gap of the motor. One of the main and most challenging design requirements was that the used material for the can should be electrically non-conductive in order to avoid excessive heat generation developed by eddy current losses in the strong motor magnetic field. This eliminated the possibility to use metal alloys or even carbon-fire reinforced polymers. After extensive qualification and testing the solution was found by using a special glass-fiber reinforced polymer (GFRP). A further requirement was to limit the wall thickness of the can. This would otherwise have a detrimental effect on motor power factor which would result in poor electrical effectiveness. The final can wall thickness achieved was 5.5 millimeters and allows a limited differential pressure over this cylindrical component. In order to achieve the full design pressure of the prototype unit of 150 barg, the pressure of the cooling system fluid of the motor stator modulates with the suction pressure by means of a simple piston-style compensator. In normal operation and as a result of smart selection of the pressure reference points, the fluid pressure of the insulation oil is slightly higher than the internal process gas pressure, thus avoiding any pressure induced migration of process gas into the closed liquid cooling system.

Separating the high speed rotor was achieved by adopting an advanced solid rotor design instead of rotor laminations. The squirrel cage is formed by copper bars and short circuit rings that are integrated in the rotor by means of hot isostatic pressing (HIP). All copper is then protected from the process gas by means of a corrosion and erosion resistant alloy cladding.

Separating the Active Magnetic bearing stator windings and sensors from the process gas is achieved by applying a thin sheet of metal welded to the bearing bracket. The pressure load on the thin sheet is supported by the stator bearing internal structure. The bearing cavities are atmospheric and cooling is arranged through heat conduction to the integral bearing bracket and housing or with additional cooling jacket around the bearing as part of the integral motor cooling system.

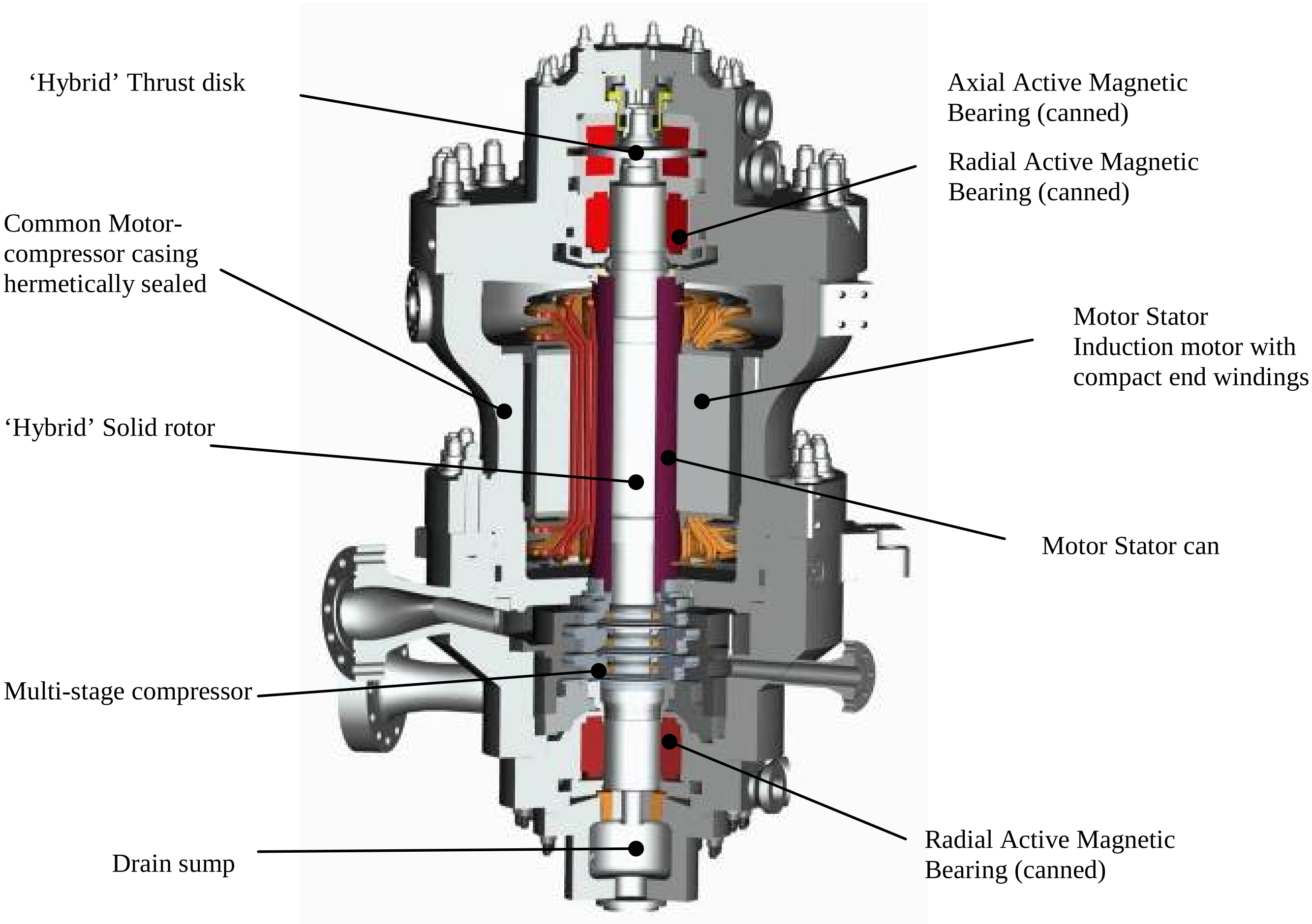


Figure 1 Hermetically sealed compressor design concept

Hermetically sealed compressor technology has been around since the early '90s with integrated motor and magnetic bearings, with a prime concept of using process gas to extract heat from motor stator windings and bearings, ref (1). Development of the fully canned hermetically sealed unit dates back to 1999 initiated by a request from Shell Global Solutions to develop a next generation compressor capable of handling untreated process gas directly received from oil production facilities and gas fields and with zero emission to the environment (i.e. no shaft seal leakages and capable for pressurized standstill). The concept had to be suitable for an array of gas contaminants ranging from water to boiling cuts, mercury, solid particles, condensate and H_2S. The particular application at NAM's Vries-4 gas gathering site near Groningen, The Netherlands falls in material category 1, with 3 MW power requirement at 12200 rpm, ref (2).

Months of extensive in-house testing preceded installation in autumn 2006 of the prototype unit. A scheduled inspection according the manufacturer's stringent Product Development Process after its first year of general operation in saturated gas confirmed that the prototype was in excellent condition. None of the components required replacement or refurbishment and the unit was re-assembled and regained its full operation three weeks later.

Figure 2 First hermetically sealed unit K-270 installed at a gas-gathering facility at Vries, The Netherlands, 2006 (Courtesy of NAM)

In the period 2010 to 2014 the unit was further extensively monitored to provide availability and reliability data for the validation milestone in the Author's company development program, see Figure 3. The unit proved its suitability for the intended applications supporting confidence in achieving the desired 5-years maintenance interval, which had been among the key performance criteria in the development of hermetically sealed compression technology. To date, the unit has run over 50.000 hours on well stream process gas.

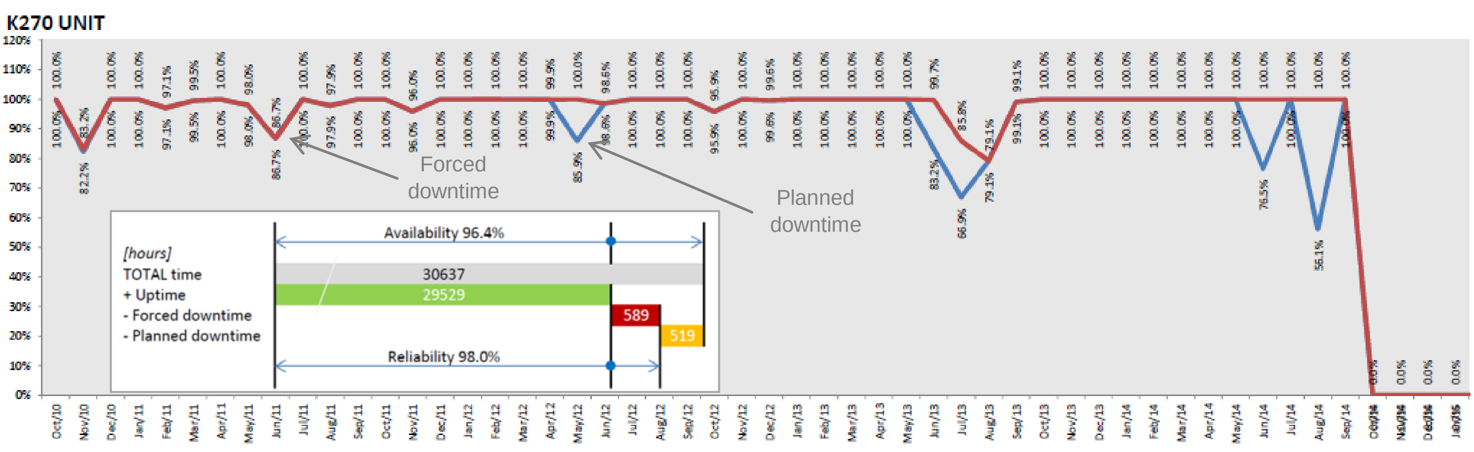


Figure 3 Availability and Reliability data for the first prototype unit, recorded over 4 years interval (Courtesy of NAM)

The first prototype revealed a number of design challenges and operational restrictions in thermodynamics and rotor dynamics that are inherent to hermetically sealed technology. One of which was the restriction of a rotor having small annular gaps running in high density fluids due to destabilizing effects. This absorbed quite a large portion of the radial bearing capacity. A product improvement development program was initiated to improve its behavior and allowing the hermetically sealed compressor to operate at higher suction densities. A second, even more compact and more advanced unit was designed and built for Statoil in 2008 and was submitted to extensive in-house FAT testing at full-load and full pressure on natural gas (PTC-10 Type 1) at the Author's company facility in Duisburg, Germany.

Figure 4 K-lab, a 5.8 MW Hermetically sealed unit for in-house testing on Natural Gas

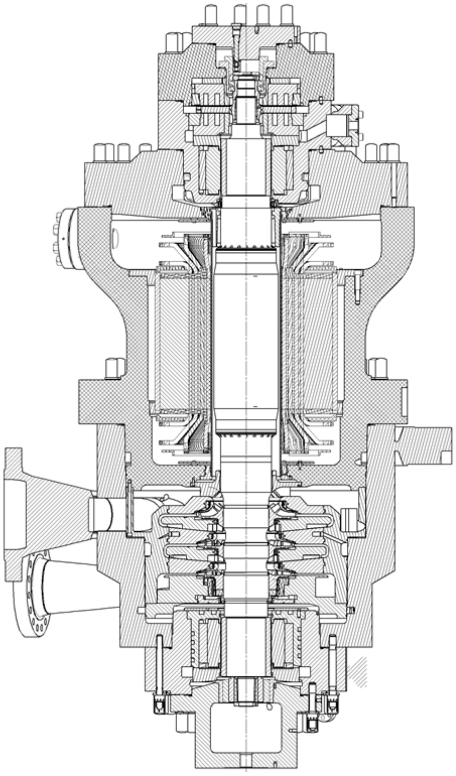


Figure 5 Cross-section of hermetically sealed compressor for Sour and Corrosive gases

The typical characteristics of the hermetically sealed design concept are:

- Vertical arrangement: allowing minimum plot plan, a limited static load on the radial bearings and single point gravity-induced drainage at the bottom
- Motor section on top, away from the hot compressor discharge and simplifying field maintenance by means of a pull-out pack concept without the need to break process lines on the compressor section.
- Non-conductive stator 'can' to protect the motor stator (patented), thin walled to limit reduction of the motor power factor and compatible to process gas, H_2S and CO_2
- Single and solid 'hybrid' rotor (patented), containing two different materials allowing corrosion resistance as well as good magnetic properties. The rotor outer surface is canned by means of alloy cladding protecting the copper components
- Solid 'hybrid' thrust disk for sour and corrosive gases (patented), containing two different materials allowing corrosion resistance as well as good magnetic properties
- Compressor discharge side on the bottom
- Two canned radial active magnetic bearings
- One canned double-acting thrust bearing on top
- Full redundant radial position sensors
- Auxiliary bearings arranged at either shaft end for easy maintenance
- Process gas cooling on all rotating sections, cooling gas is extracted after the first impeller
- Liquid closed loop cooling circuit for the motor stator
- Compressor inlet plenum area is designed to handle significant amount of liquid carry-over (targeting down to 98% gas volume fraction), which is partly separated and drained to the bottom collector and partly carried along with the main flow depending on droplet size
- All current units are of similar compact design with a power capability of 6 MW, maximum speed of 12200 rpm and a design pressure of 150 barg. The same technology with motor and compressor in between two radial bearings targets further extension up to maximum 12 MW

- Designed for prolonged operation without intervention or maintenance by eliminating low-MTBF components like dry gas seals and applying state-of-the-art technology i.e. high grade material selection and maintenance-free magnetic bearing technology. Typical advised spare parts include an auxiliary bearing set and a pump strainer, all of which are easily accessible without major disassembly

ADVANCED TECHNOLOGY FOR SOUR AND CORROSIVE GAS

Composition of the gas determines its corrosivity. In natural gas the corrosive components are CO_2, H_2S and chlorides in water which is in the gas. When the partial pressure of H_2S is larger than 0.0034 bar, the gas is considered as sour gas. Materials which can be used in this gas are defined in ISO 15156 (NACE). With increasing partial pressure of CO_2 and H_2S, the corrosivity of the gas increases. Relative humidity of the gas contributes substantially to the corrosivity. If the relative humidity is less than 40% gas can be considered dry. Carbon steels can be used without danger of extensive corrosion with high CO_2 partial pressures. However if the relative humidity is high up to 100% the situation drastically changes. Above 1 bar partial pressure CO_2 carbon steel is no longer suitable from corrosion resistance point of view. In such a case, corrosion resistant alloys have to be used in accordance with ISO 15156-3.

The gas for the subject upstream compressor contains considerable amounts of CO_2 in combination with H_2S, see Table 2. Moreover there is water vapor present in the gas. The gas therefore has to be classified as Category 3. This means that the material used for the compressor has to withstand corrosive gas. Carbon steels do not have adequate corrosion resistance to wet CO_2. Therefore they were ruled out. There are several corrosion resistant material to choose from. All these are listed in ISO 15156-3. The material solution has to be also economical. For the last 25 years 13% Cr steels have been the most cost effective solution to protect against sweet wet CO_2 corrosion, ref (2).

Table 2 Component levels in the gas composition of the subject upstream compressor for Saudi Aramco

Component	Water vapour	Nitrogen	Carbon Dioxide	Hydrogen Sulphide	Methane + rest
Amount %	3.8 %	0.08 %	95.0 %	0.67 %	Rest
Partial pressure	0.94 bar	NA	7.79 bar	0.05 bar	NA

Hybrid Rotor Design

It is clear that the rotor must also satisfy magnetic requirements in order to function as a rotor of the motor. This means that the material of the shaft has to be ferromagnetic. Since corrosion resistant alloys (CRA) do not meet the magnetic requirements of the motor, one solution is to fabricate a *hybrid* shaft out of two different materials. One which will satisfy the magnetic requirements and another which will have sufficient corrosion resistance.

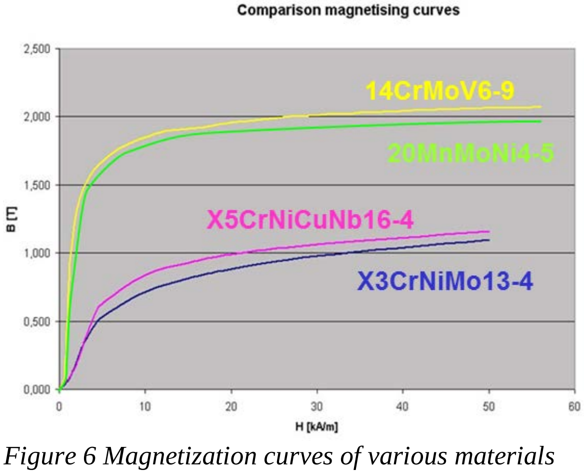


Figure 6 Magnetization curves of various materials

The solution was to weld two different materials to fabricate the shaft. Material of choice for magnetic properties as well as mechanical properties was 20MnMoNi4-5, see Figure 6. This material will not meet the corrosion requirements and should therefore not be in contact with the process gas. Material chosen for corrosion resistance was X3CrNiMo 13-4, as indicated in a schematic drawing of the composite shaft in Figure 7. The shaft was fabricated in two sections because of size restrictions of the post weld heat treatment ovens and other processing equipment. The section made of X3CrNiMo13-4 forms the lower compressor part of the rotor. The second section consists of a central core of 20MnMoNi 4-5 welded on both sides to parts made from X3CrNiMo 13-4. Slots were machined into the core for fixing copper bars and short circuit rings which formed the motor rotor. The copper bars were attached using an established process of hot isostatic pressing (HIP). Finally the first and the second parts were welded together forming the complete shaft and was then final machined. The corrosion resistance of the motor rotor was still not adequate for the corrosive gas of the upstream application. To overcome this, the rotor was cladded with thermally sprayed Alloy 625 and finally ground to finish. This completely protected the rotor from the corrosive gas. Material of choice for the impellers from the point of view of mechanical strength was X5CrNiMo 17-4 which has adequate corrosion resistance.

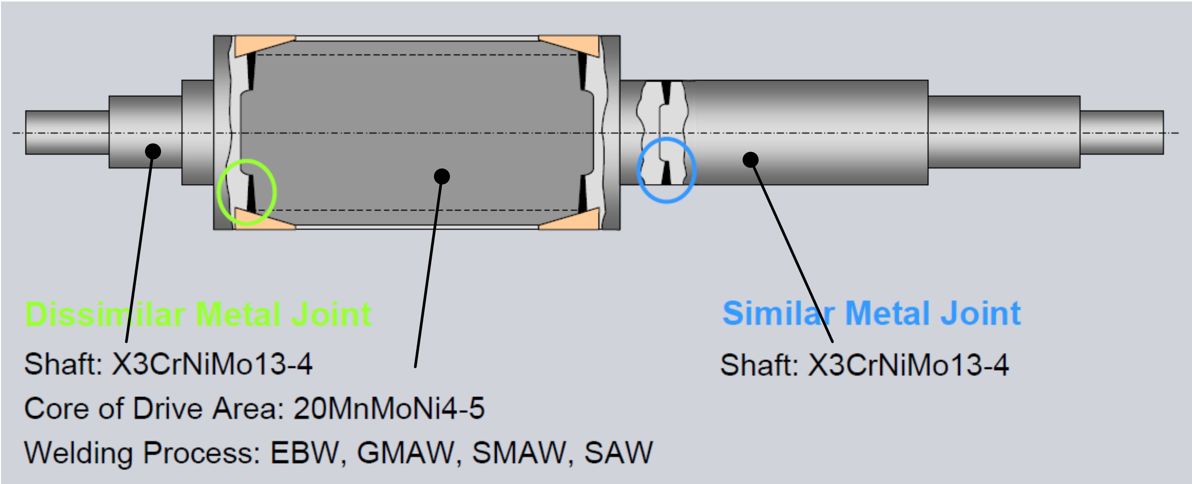
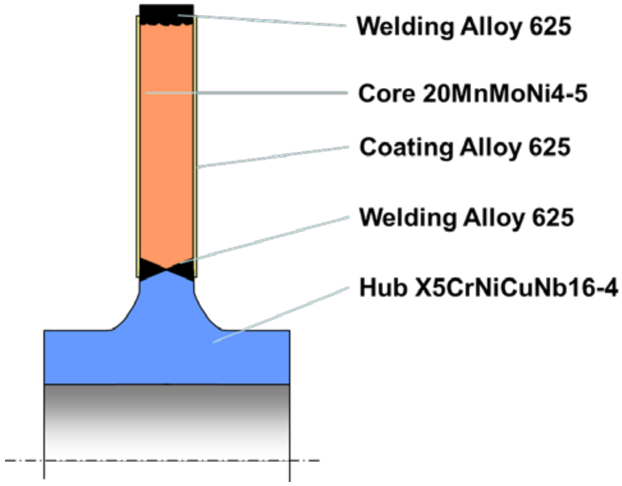


Figure 7 Hybrid rotor for motor-compressor

Hybrid Thrust Disk

The shaft contains a thrust disk which magnetically suspends the shaft. This means that this thrust collar has to be ferromagnetic. The thrust disk was manufactured by a ferromagnetic core, covered on the outside diameter with weld metal of Alloy 625 and on the inner diameter with a ring out of X5CrNiCuNb16-4 jointed with filler metal of Alloy 625. In order to protect the material surfaces from corrosive gas, cladding of Alloy 625 was applied using a thermal spraying process. Finally this solid disk was shrunk on the shaft.



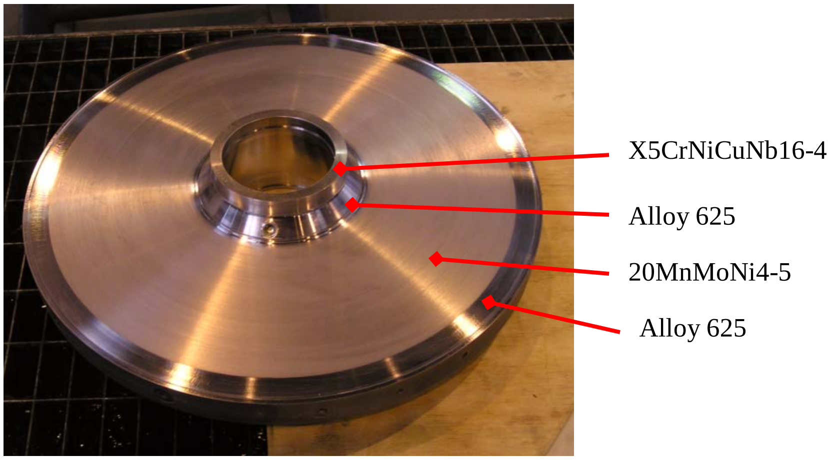


Figure 8 Hybrid thrust disk prior covering with thermal sprayed Alloy 625

Motor stator 'can'

One particular characteristic of the hermetically sealed compressor concept is that a part of the pressurized process gas flows through the air gap between motor rotor and motor stator for extracting the heat because of close clearance friction losses. In order to protect the electrical insulation of the motor windings a fiber-reinforced polymer cylinder, 'can', fills part of the air gap and is in contact with the process gas on the inner surface and the motor cooling fluid on the outer surface. On the outer surface a small gap of 1 mm remains towards the motor stator, on the inner surface a larger air gap of 9 mm is provided to allow sufficient cooling gas over the motor rotor whilst maintaining a good overall motor power factor. The sealing at both ends consist out of a double T-shaped explosion compression resistant sealing element.

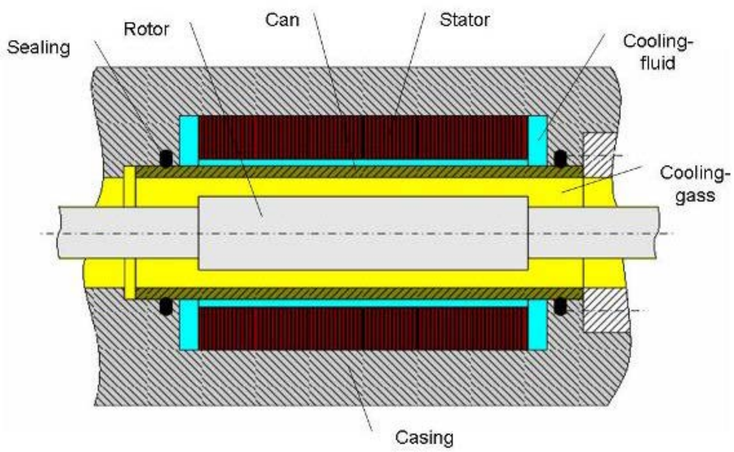


Figure 9 Schematic set-up of the motor section with integrated non-conductive separation 'can'

Figure 10 Non-conductive separation motor stator 'can' ready for installation in the hermetically sealed compressor

Functional Requirements

The main function of the stator can is to protect the motor electrical windings from the untreated process gas. This implies a large array of functional requirements:

- Minimum wall thickness, finally reaching 5.5 mm
- Minimum radial deflection to avoid heavy radial load on the motor stator laminations
- Dimensional stability on sealing areas to avoid any leakage of cooling fluid into the process gas
- Compatible with Hydrocarbon gases, contaminants and cleaning agents
- Compatible with H_2S, CO_2 and H_2O (inner surface)
- Compatible with insulating cooling oil (outer surface)
- Wear resistant to solid particles and droplet erosion
- Non-conductive to avoid excessive heat built up due to eddy current losses
- Minimum permeation of process gas (pressure or chemically induced)
- Temperature resistant up to 150ºC

Qualification testing

After selection and tests with various materials and resins, the final selected glass-fiber reinforced polymer was conducted to a rigorous qualification program adopted on both small scale samples as on full-size can including:

- Sand erosion testing (ASTM G76)
- Explosive decompression testing
- Chemical compatibility testing
- Membrane permeation testing (gas tightness)
- High temperature testing
- High differential pressure testing
- Internal overpressure testing (destructive)
- External overpressure testing (destructive)
- Long term test with cycling differential pressure
- Long term 1000 hour pressure testing
- Acceptance testing on maximum design parameters
- Compatibility tests on H_2S, CO_2 and H_2O

The extensive testing program included small scale and full scale testing and yielded a qualified material and manufacturing process using a glass-fiber reinforced polymer.

Active Magnetic Bearing system

Magnetic bearings at the top and the bottom were supplied by an external supplier and subsequently installed in the integral bearing brackets. For the required corrosion resistance and magnetic properties, rotor laminations were made of stainless steel. For auxiliary bearings, a sleeve of Alloy 625 was used for corrosion resistance. Both Magnetic bearing and auxiliary bearing stators were canned by a sheet of Alloy 625 for corrosion protection from corrosive gas.

Figure 11 Canned radial and axial Active Magnetic Bearings integrated in the bearing housing and with high pressure power and signal feed-through

Motor

The integrated motor is a high-voltage, high-speed AC, asynchronous solid squirrel-cage motor paired with a variable frequency drive (VFD). In order to enable a compact two-bearing compressor with motor and compressor section in-between bearings, the motor stator design required stringent space limitations. Therefore, the stator was built from wiring rather than copper bars in order to allow a very compact 90 degree end winding concept, hence to limit the overall bearing span. The coils and the finished windings are insulated with the patented MICALASTIC® vacuum pressure-impregnation system (Class F). In-between the two winding layers PT100 temperature sensors are placed in spots that best represent the maximum winding temperature.

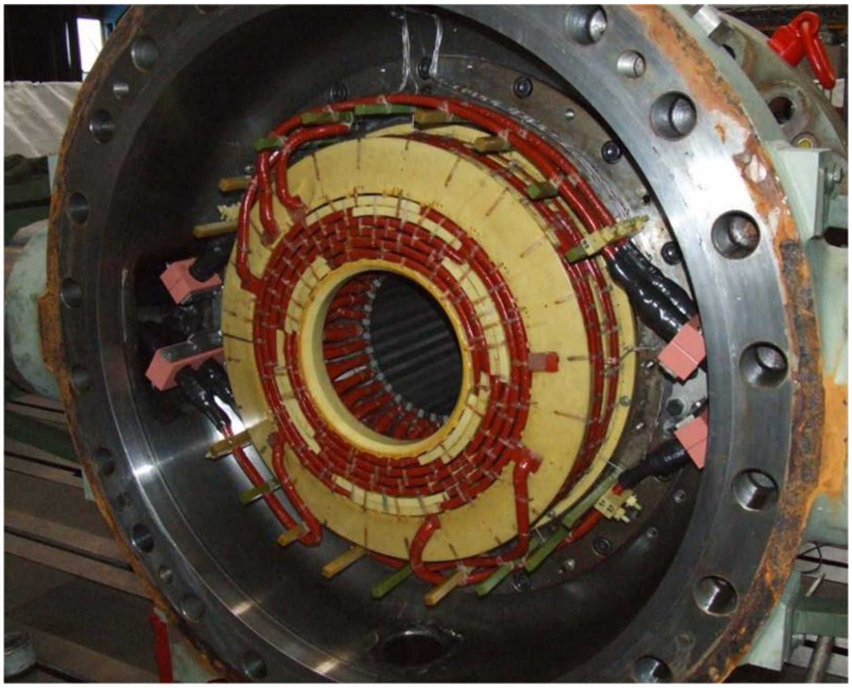


Figure 12 Motor stator in excellent preserved condition after one year of operation on a gas production asset in The Netherlands (2007, Vries One Year field trial)

Lateral Rotordynamics

Applying only two radial bearings obviously is a drastic simplification of the overall concept as it eliminates the need for a mid-span bearing that would be hard to access. On the other hand the integration of an induction motor between the bearings increases the overall bearing span compared to that of a comparable conventional compressor with dry gas seals. For instance the bearing span of the prototype unit is 165% of the bearing span for an equivalent 5-stage, traditional compressor equipped with dry gas seals and oil-lubricated bearings. However, the lateral characteristics of the hermetically sealed compressor are comparable to a conventional compressor. The key to this is the solid motor rotor in combination with very compact stator end winding design. This results in relatively robust rotor geometry with a slenderness ratio (bearing span/average shaft diameter) of 9:2 and total rotor mass of 1619 kg.

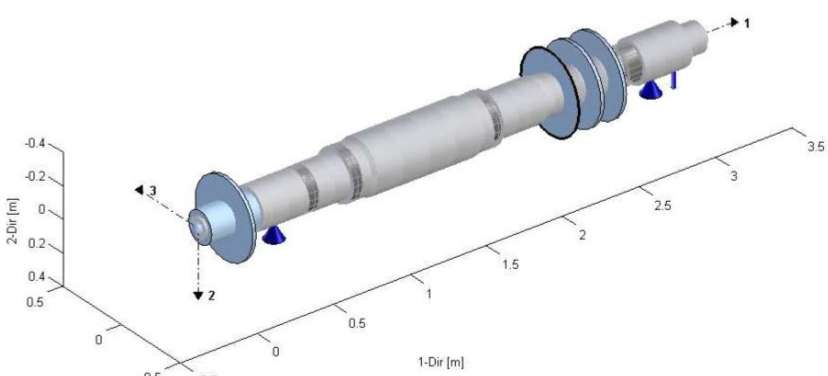


Figure 13 Model of the motor-compressor rotor for the subject compressor

Since the complete motor-compressor is supported in a steel frame, the characteristics of this supporting structure need to be taken into account in the lateral rotor response analysis. The frame is modeled in form of a spring bearing support, i.e. additional spring-, damping- and mass-coefficients are added at the bearing nodes. According to Author's companies experience this is an adequate way to model the frame flexibility and to gain reliable calculation results. The closed loop rotor dynamic analysis is required for this type of combined mechanical and control systems confirmed the feasibility of this concept. It assesses both controllability as well as observability of the rotor. In a first approach a linear digital controller is used to guarantee stability of the rotor-dynamic system. In a next step the basic controller design can be modified by some nonlinear features to optimize the rotor-dynamic stability, vibration amplitudes and the behaviour at over-critical speeds. There are several possibilities of non-linear controller designs that can be used.

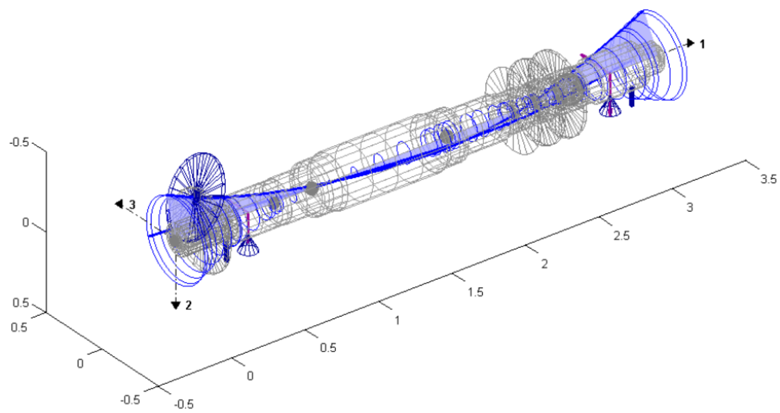


Figure 14 Rotor first bending mode

Motor cooling system

A dedicated closed loop cooling system is used to extract the heat generated by electrical losses in the motor stator whilst maintaining a small differential pressure across the separation can. A liquid for heat transfer is much more effective than a gas due to the higher inherent thermal conductivity. Furthermore, the used bio-degradable oil as coolant has excellent insulation properties to increase short circuit tolerance within the motor, contributing to high overall machine reliability over long periods of uninterrupted operation. The cooling system consists of 3 main components: a pump, a cooler and a pressure compensator and is therefore characterized as a "forced flow cooling system", see Figure 16. The screw type positive displacement pump circulates the cooling medium and is a seal-less type where the motor drives the pump via a magnetic coupling. In this magnetic coupling a metallic can ensures a reliable hermetic sealed pump design. In normal operation the pump is low loaded with a flow rate of 423 l/min and a differential pressure of only 1.5 bar (shaft power 2.2 kW). In the subject application a redundant (2x100%) pump configuration has been selected with automatic change-over functionality. When the main pump is in operation, maintenance or replacement of the spare pump unit is possible. The redundant configuration of this low loaded, hermetically sealed pump arrangement ensures a high system reliability and availability.

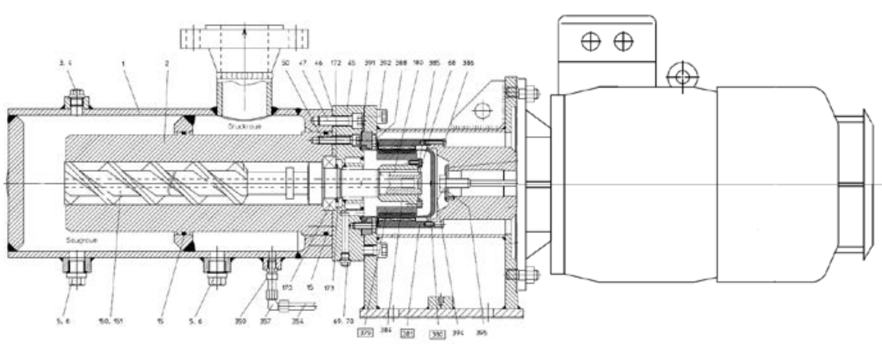


Figure 15 Cooling system pump (courtesy of Leistritz Pumpen GmbH)

The cooler in the subject upstream compressor application is an air type cooler. To maintain a limited pressure differential over the motor stator can, a pressure compensator is used to balance the coolant pressure with the compressor suction gas pressure. Its second function is to keep the gas and coolant separated and allow for thermal expansion of the cooling medium. The coolant is an insulating oil Midel 7131® entering the motor stator area extracting the heat while flowing from bottom to top through a large number of small bores in close proximity to the stator winding slots. It has also reasonable thermal properties; low expansion, heat transfer properties comparable with mineral oil and a high fire point of 322 °C.

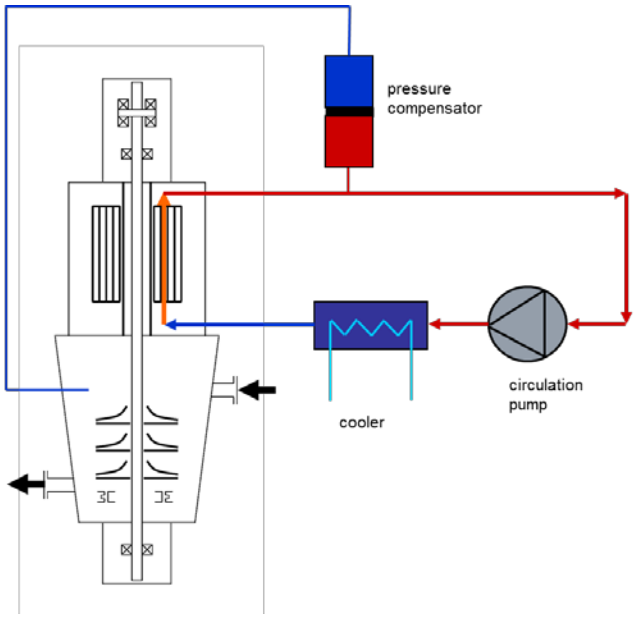


Figure 16 Motor cooling system using insulation oil under pressure, modulating with suction pressure

Rotor Cooling system

When a rotor is running at high rotational speed in high density gas, there will be a significant amount of heat generated due to close clearance frictional losses. Depending on speed and suction density this can reach values in the range of 20 to 400 kW. Such heat is generated in all close clearance gaps inside the compressor unit as well as heat generated by electrical rotor losses:

- Motor rotor gap: between rotor and separation can
- Radial active magnetic bearings: between rotor and AMB can
- Axial active magnetic bearing: between thrust disk and AMB can
- Auxiliary bearings: between sleeve and bearing stator
- Motor rotor: electrical losses

All of these losses need to be extracted by a gas cooling system. The cooling gas is untreated process gas taken off after the first impeller and distributed by dedicated flow lines through the relevant components. All gas flows return to the compressor suction and mix with the main process flow, slightly heating this up. The circulation of cooling gas contributes to the overall losses that are inherent to hermetically sealed compressors. Since the flow resistance in these components is relatively low, the pressure difference between first impeller discharge and the suction pressure (which drives the flow) would generate a relatively large flow. Therefore the flow distribution is controlled (restricted) by flow orifices in each supply line.

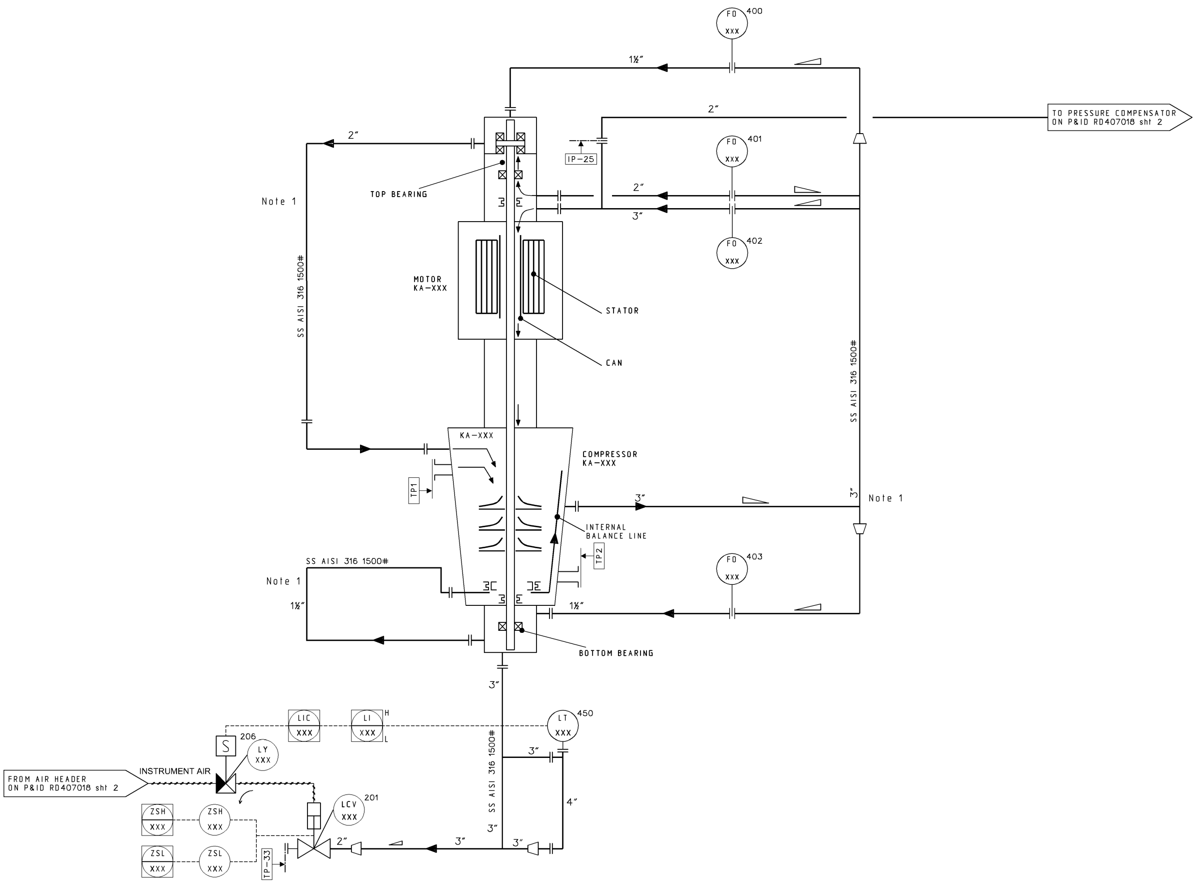


Figure 17 Rotor cooling system using untreated process gas extracted after the first impeller

UNIT TESTING

Similar to conventional centrifugal compressor units the hermetically sealed compressor is tested on component and subassembly level during its manufacturing and assembly process. Typical tests are material NDE, hydrostatic testing of pressure containing parts and overspeed testing of rotating elements. A specific example of a manufacturing test for an hermetically sealed compressor is a function component test of the GFRP stator can which is tested against internal as well as external overpressure well beyond the safeguarding (alarm/trip) settings during normal operation of the compressor. A specific test during assembly of the unit is an extensive Helium leakage test of the bearing bracket assemblies where the active magnetic bearing stators are seal welded into the bearing brackets by means of thin metal cans.

After full assembly of the compressor unit a series of unit tests are performed in the manufacturer's shop in order to generally proof suitability for the field conditions to which it will be exposed after shipment to site and specifically to show that contractual guarantees are met.

The main data of the motor-compressor train are given in the listing below.

- Idle speed : 950 rpm (10%)
- Minimum operating speed : 2850 rpm (30%)
- Rated operating speed : 9500 rpm (100%)
- Maximum continuous speed : 9975 rpm (105%)
- Trip speed : 10165 rpm (107%)
- Overspeed : 10973 rpm (115.5%)
- Motor rating: 5.8 MW

Figure 18 Hermetically sealed compressor for sour and corrosive gas being installed for PTC-10 Type 1 testing in Hengelo, The Netherlands

Mechanical running test

During the mechanical running test at full load conditions vibration levels and internal clearances are verified against API 617 annex F. Prior to running tests, internal clearances are measured using the active magnetic bearings to offset the rotor. Shifting the zero rotor position in the AMB moves the rotor from its actual center position towards the outer wall within the mechanical clearance of the machine. When this clearance is exceeded the AMB electrical current shows a step increase. During the full load running test all auxiliary systems are functionally tested: motor stator liquid cooling system, rotor gas cooling system and electrical and control systems. Compressor unit rotor vibration data have been recorded where API 617 annex 4F is referenced for acceptance criteria (F.7.6). It states that the maximum rotor movement relative to the auxiliary bearing center is 0.3 times the minimum radial clearance in the radial auxiliary bearing. Figure 19 shows recorded vibration data in top and bottom radial bearing during shop testing at maximum operating speed and full load conditions. Direct radial vibrations recorded are stable and reach maximum 50 micron peak-peak at the top bearing and maximum 30 micron peak-peak at the bottom bearing. During these conditions the rotor center was running in an off-center measured position of 19 micron at the top and 22 micron at the bottom. Hence, total rotor movements (AC+DC) relative to the bearing geometrical center was 25 (0-peak) +19 (off-set) = 44 micron at the top bearing and 15 (0-peak) + 22 (off-set) = 37 micron at the bottom bearing. With a minimum radial auxiliary bearing clearance of 0.26 mm the API acceptance criterion is 0.3*260=78 micron. It can be concluded that the total rotor movements relative to the bearing center are well within API recommended values.

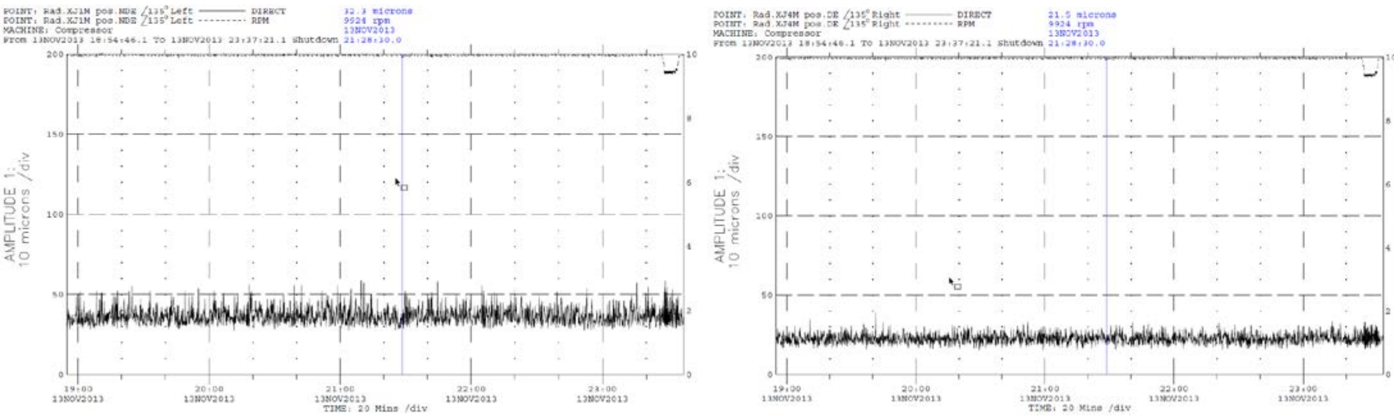


Figure 19 Radial vibration trend data during mechanical running test at full load conditions on Top bearing (left) and Bottom bearing (right)

Driver heat run test

A heat run test at specified load of 4.3 MW is performed to determine the rated temperature rise of the motor. The stator winding temperatures should not exceed 155°C (Class F temperature rise) at stable condition, defined as maximum temperature variation of 2°C per hour. Measured motor winding temperatures are reflected in Figure 20 showing these are well within the maximum allowed value (max measured value is 74°C). The bearing temperatures reached a maximum stabilized temperature of 117°C at the hot end (bottom) of the machine after running 4 hours at maximum continuous operating speed. For the magnetic bearing with a trip setting of 145°C this is considered acceptable.

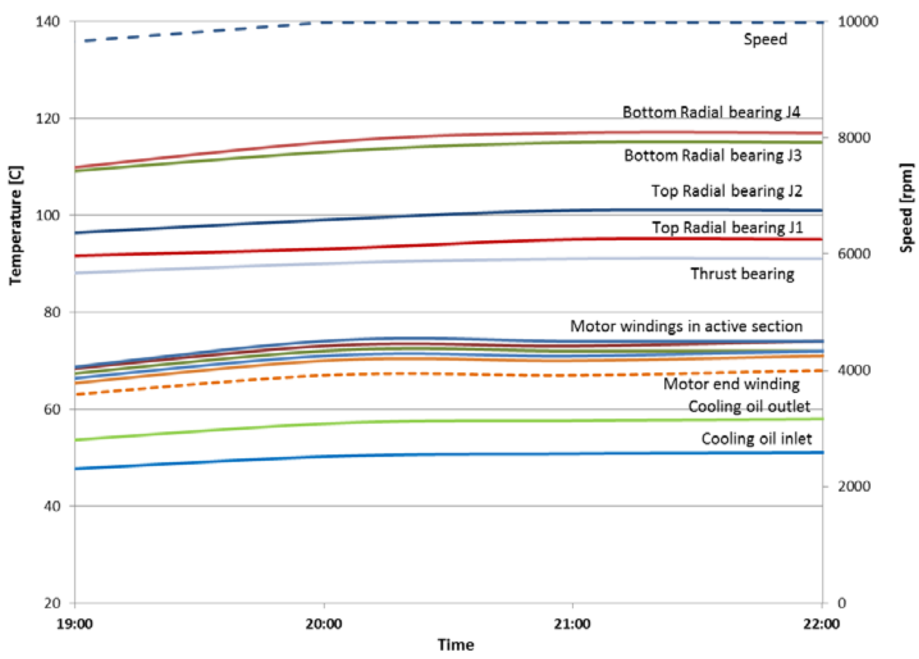


Figure 20 Motor and bearing winding temperatures and cooling medium temperatures during heat run test at full load 4.3 MW

PTC-10 type 1 testing

In order to verify the thermodynamic performance of the compressor the unit has been closed loop tested in accordance with ASME PTC-10 type 1 on equivalent an process gas mixture. For this CO_2-compressor a mixture of 92.5% CO_2 and 7.5% N_2 is used. Acceptance criteria are defined in API 617: for a variable speed machine the head and capacity shall have a zero negative tolerance where speed correction is allowed in order to achieve the specified performance. In respect to power a contractual end user

requirement allows the guaranteed mechanical power being exceeded for maximum 1% where API 617 typically allows 4%.
The test results show that after a speed correction of +1.8% in order to meet the predicted mass flow and polytropic head, the total motor power is 2.9% more than predicted but still 1% below the guaranteed value.

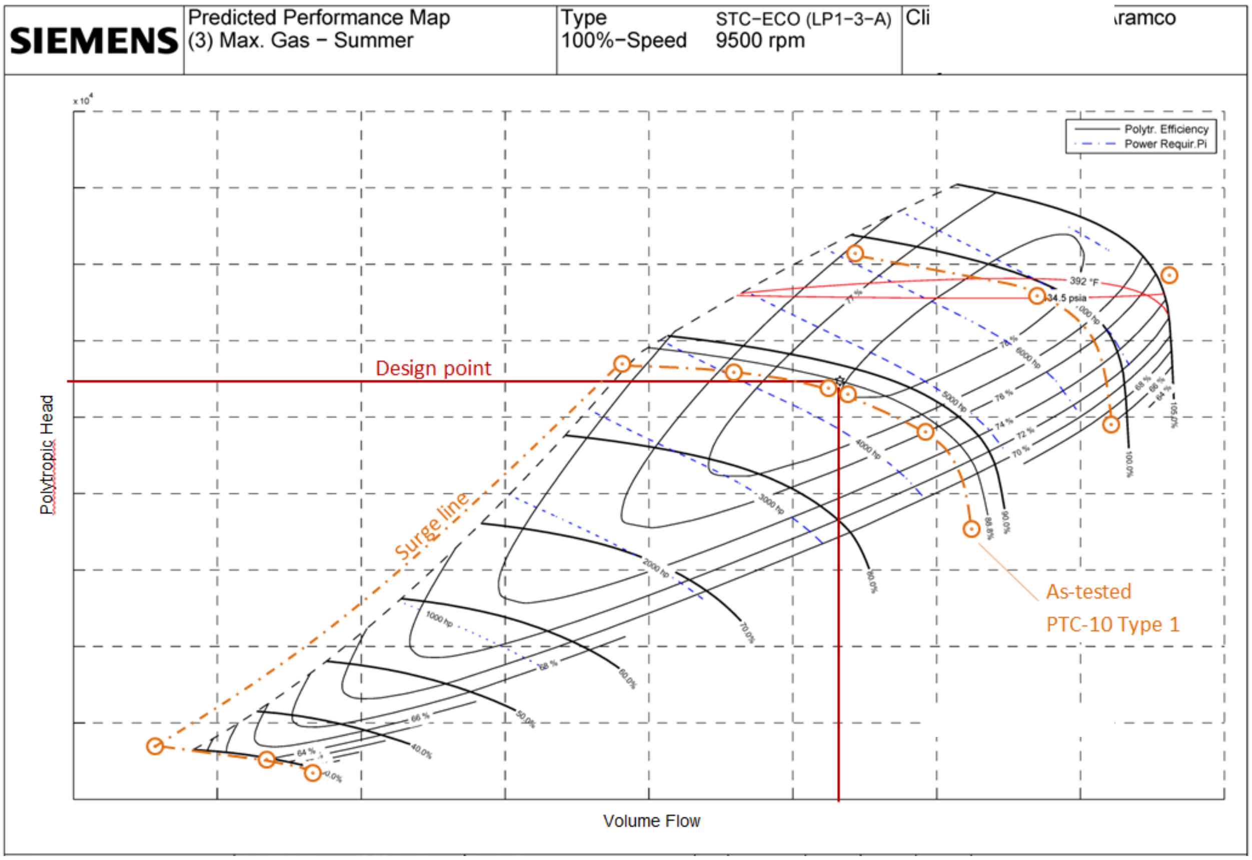


Figure 21 CO2 compressor performance characteristic, FAT test

Power

Due to the specific design of the hermetically sealed compressor concept with its integrated motor, power and efficiency cannot be compared on a one-to-one basis to conventional design centrifugal compressors driven by an external driver. The suction density gas in the small gaps between high-speed rotating shaft and static compressor internals result in significant windage losses cause by shear. The heat developed in these gaps is removed by an internal gas recycle flow where gas is taken from the first impeller discharge, routed through the internal gaps in the motor and bearing sections and ultimately reinjected upstream the first impeller. This causes an additional temperature increase to the main gas flow. The first impeller head is used as driving force for this ‘cooling gas’ recycle flow. In the compressor thermodynamic design calculations this recycle flow and re-heat effect is implement analogue to the balance drum leakage recycle flow. However, when deriving gas power from enthalpy change based on compressor inlet and outlet gas temperature and pressure measurements, the measurements shall be corrected for the close clearance friction losses in motor and magnetic bearing cavities in order to be able to compare power and efficiency of a hermetically sealed compressor to conventional compressors and their API-617 requirements.

In order to illustrate this, a Sankey diagram in Figure 22 shows how the actual required motor power *P_gas_total* is the sum of (conventional) gas power *P_gas* (including typical stage losses and balance drum flow) and all internal losses caused by disk friction, electrical rotor losses and cooling flow recirculation that are inherent to hermetically sealed compression. As power is determined from the enthalpy rise over the main nozzles with all of these losses included, it is required to extract the disk friction, electrical and cooling flow losses in order to determine ‘classical’ polytropic efficiency. This is a methodology that is not covered in API-617 and PTC-10.

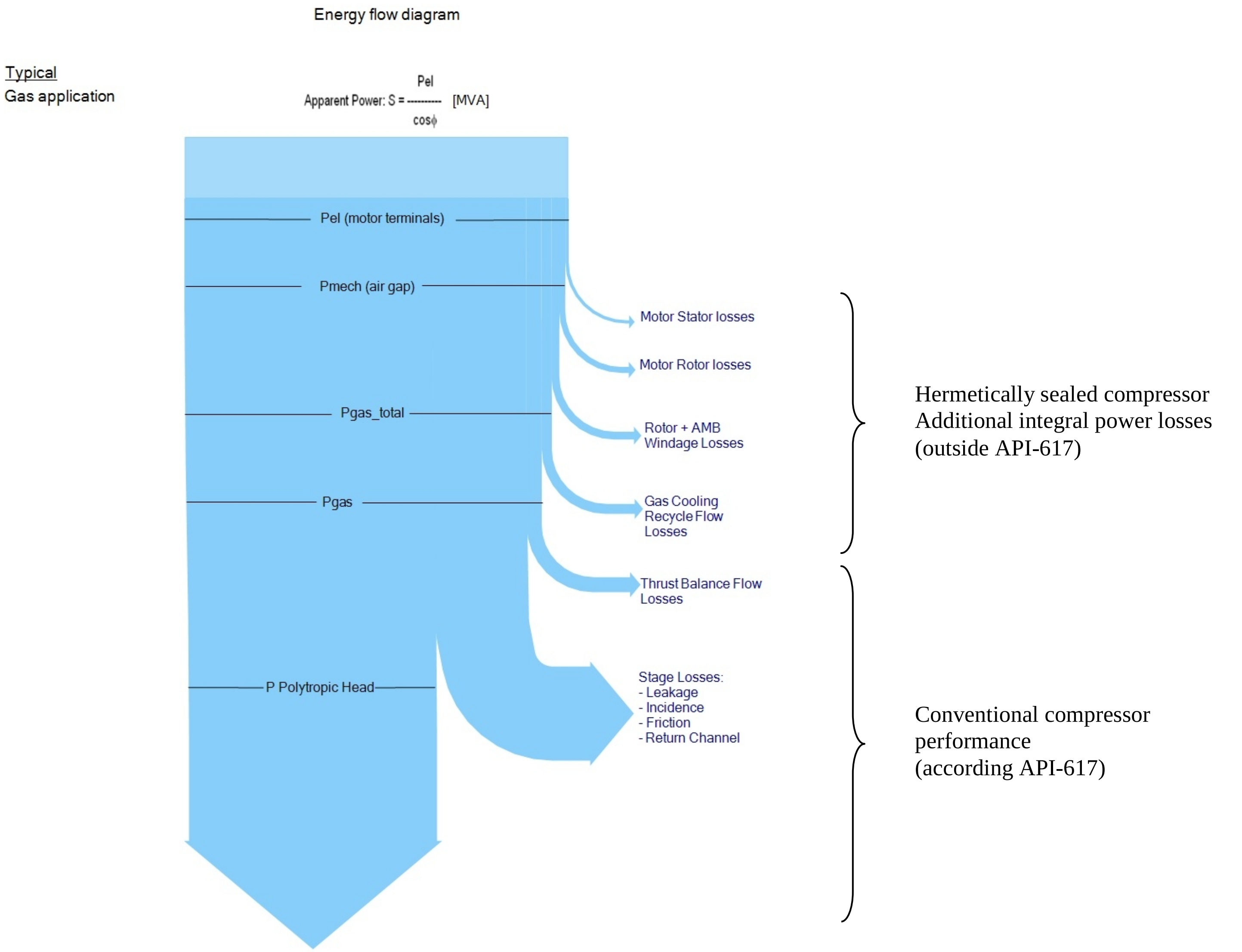


Figure 22 Sankey diagram visualizing breakdown of internal power losses typical for hermetically sealed compressors

Emergency landing test

During operation the single shaft motor/compressor rotor is levitated in an active magnetic bearing (AMB) system consisting of 2 radial AMB's and 1 axial AMB at the top of the vertical arrange unit, a 5-axis system (2 for each radial bearing and 1 for the axial bearing). The AMB system is powered from the local grid via an uninterruptable power supply (UPS). In case the main grid would fail the UPS batteries provide sufficient power to ensure a safe levitated rundown of the motor/compressor rotor. In the ultimate event that also the UPS would fail, an auxiliary or backup bearing system (with ball or sliding bearings) will ensure a safe non-levitated rotor run-down to standstill. These auxiliary bearings are also used in case of transient overloading of the AMB system due to i.e. significant liquid content in the process gas. In the extreme case of a non-levitated rundown in the auxiliary bearing system, an incorrect designed system could result in unstable rotor rundown where in the extreme case a backward whirl of the high speed running rotor could lead to big rotor excursions and consequently to damage of the compressor internals. To proof the ability of the auxiliary bearing system for a safe rundown of the rotor a so called emergency landing test is executed where rotor is landing in the auxiliary bearings at full speed, initiated by manual de-levitation of the rotor in all 5 axes. Rotordynamic characteristic are being recorded during the test for further analysis. Auxiliary clearances are checked before and after the test concluding functionality and resistance to wear.

Due to the high circumferential speed and weight of the compressor rotor the auxiliary bearings are exposed to high loads during an emergency landing and rundown. Consequently the auxiliary bearings can take only a limited number of landings at full speed (typically a guaranteed minimum of 5). In order to test the durability of the used auxiliary bearing system a qualification test has been executed on a previous build hermetically sealed compressor. In Siemens test center 5 sequential landings from maximum continuous speed (9975 rpm) to zero speed have been successfully applied on the test machine. Although bearings clearances increased during the several emergency landings, the system was still able to run the rotor down is a stable and safe way at the 5th landing. After inspection the pads and sleeve were in very good condition and shows basically only some "polishing" effects on the contact surface. On the subject gas compressor one emergency landing was performed as part of the overall testing program, Figure 23.

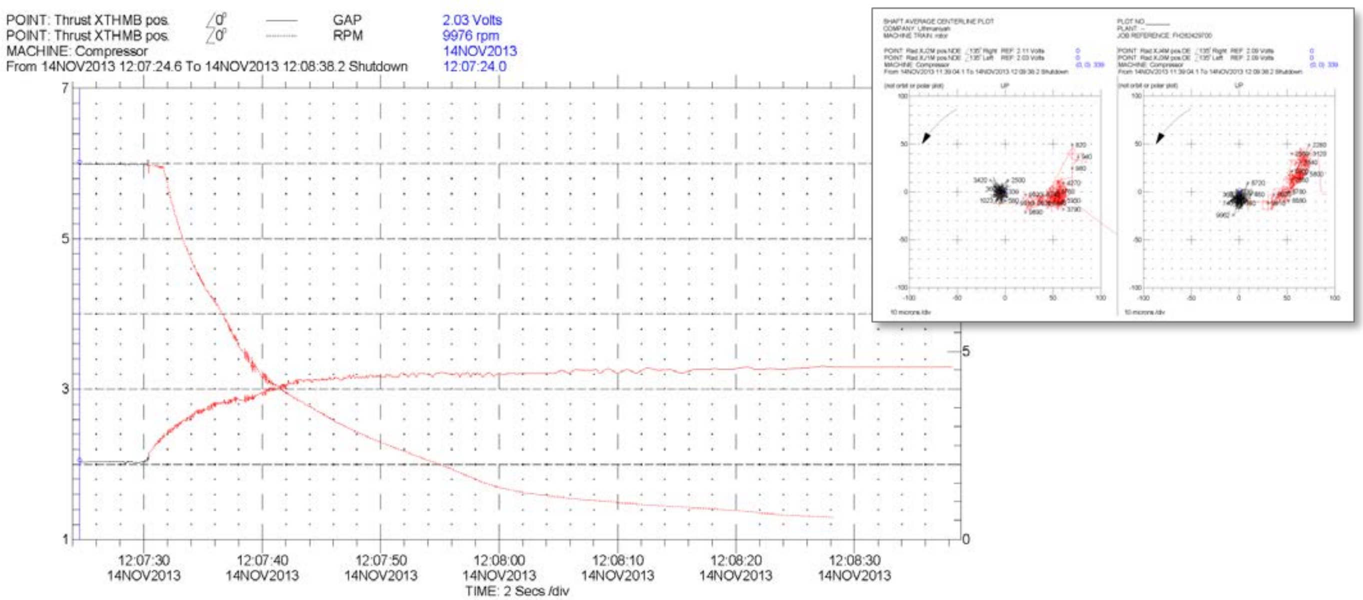


Figure 23 Axial movement and radial excursion of shaft centerline upon an Emergency Landing Test (ELT) with active magnetic bearings de-levitated from full-speed 9975 rpm

FIELD OPERATION

The hermetically sealed compressor was successfully installed at the Gas-Oil Separation Plant and commissioned on May of 2015. As such design do not use mechanical seals, lubricated bearings and their associated systems, several timely commissioning activities necessary with traditional compressors such as bearings and seals check, lube oil system site piping, cleaning and flushing were eliminated. In addition, traditional field alignment requirements between compressor, gearbox and motor as well as skid leveling were obviously eliminated since the motor and compressor are all integrated in one rotor. The equipment came packaged and needed only cable and piping connections. This reduced the commissioning time significantly with several weeks as opposed to typical pre-commissioning and start-up for classic compressors.

The major commissioning activates that were performed was compressor surge testing at several speeds for field tuning of anti-surge controller. This was done in order to ensure that surge control margin is set and configured accurately for each speed. Moreover, the magnetic bearing control system field tuning was minimized as it was fully tested and tuned during the compressor full-load factory testing.

Overall, the compressor has been running successfully with no major issue or trips since commissioning. At this writing, after almost one year of operation mid 2016, the unit accumulated around 7000 running hours. Plant commissioning and installation activities limited the availability of the unit in August and September 2015, see Figure 24. A wiring connection issue on the unit package junction box caused temporary signal loss of the speed sensor reducing availability and reliability recording in October 2015. So far, the compressor has been running around the 100% speed and 50% of rated power due to compressing low Molecular Weight gas compared to the expected range of gas composition. Currently, the compressor is in partial recycle operating mostly at the upper left corner of the operating map. This is expected as the process gas flow and Molecular Weight were anticipated to gradually increase over the facility's production life and the compressor was designed to accommodate for that. The variation is substantial as they will rise by ~35% and ~66% respectively. The initial conditions are the lighter gas and lower flow, which obviously require lower power (50% load) reached so far. The gas handled is saturated associated gas and has shown gradual increase in molecular weight as expected, with the latest analysis having roughly 80% HC and the rest are CO2, H2S and others. It falls within Category 3 (Sour & Corrosive) conditions in table 1.

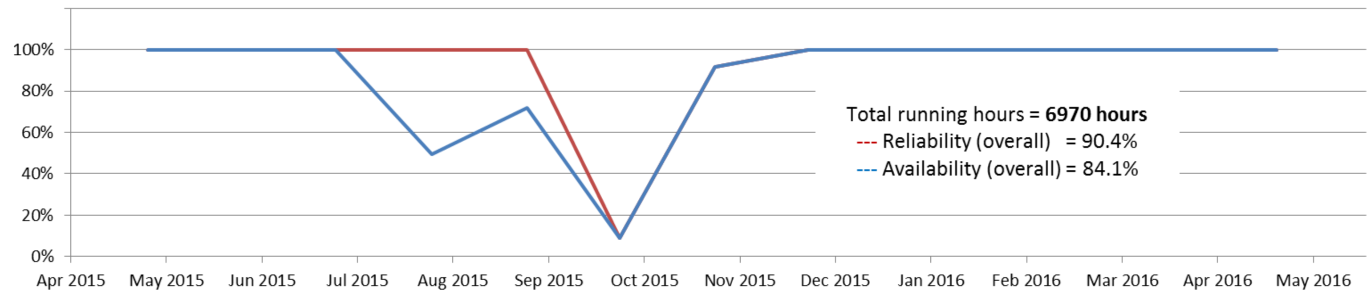


Figure 24 Availability and Reliability data for the CO2 compressor unit, recorded over 1 year interval

CONCLUSIONS

Hermetically sealed, integrated high-speed motor-compressors offer considerable advantages to compression applications. Examples of benefits to operators include wide operating flexibility, significant reduction in instrumentation, significant reduction in installed space and ease of installation and commissioning. Moreover, the operational reliability and safety are improved since hermetically sealed designs eliminate dry gas seals and lubricated bearings along with their supporting auxiliary systems, being the highest source of unscheduled shutdowns.

However, the integration of the electric motor and magnetic bearings inside the compressor internals has one major disadvantage: it would expose the vulnerable electrical insulation systems to process gas. Sour and corrosive applications require the most advanced technology to enable reliable and efficient operation of hermetically sealed designs. The subject hermetically sealed compressor can handle corrosive gases due to its 'canned' technology and 'hybrid' rotor and thrust disk design. The unit was designed and manufactured for an upstream gas processing project in the Kingdom of Saudi Arabia. The entire package was put through an extensive test program including full-pressure, full-load testing to ASME PTC-10 Type 1 and an emergency landing on the auxiliary bearing system. Determination of 'classic' polytropic efficiency on such hermetically sealed units requires a new methodology due to its inherent internal losses that are not covered in current codes and guidelines.

Upon successful completion of the test at the OEM's facility in the The Netherlands, the unit was shipped and installed in the East part of the Kingdom of Saudi Arabia and started operation in June 2015.

NOMENCLATURE

AMB = Active Magnetic Bearing
CRA = Corrosion Resistant Alloy
DGS = Dry Gas Seals
STC = Siemens Turbo Compressor
ECO = Electrical, Canned & Oil-free
GFRP = Glass Fiber Reinforced Polymer
HIP = Hot Isostatic Pressing

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